NASA TECHNICAL NOTE

NASA TN D-3672

NASA TN D-3672

N66 38143

(ACCESSION NUMBER)

19

(PAGES)

(NASA CR OR TMX OR AD NUMBER)

(THRU)

1

(CODE)

33

(CATEGORY)

FACILITY FORM 602

GPO PRICE $ ______

CFSTI PRICE(S) $ 1.00

Hard copy (HC) ______

Microfiche (MF) .50

ff 653 July 65

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • OCTOBER 1966

NASA TN D-3672

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SUMMARY

A change in buoyancy on a forced convective boiling system was effected by reversing the flow direction from upward to downward. Critical heat flux data were obtained in both directions with liquid nitrogen flowing through a 0.505-inch-inside-diameter by 12-inch-long, resistance-heated, instrumented tube. System pressure was varied from 50 to 240 pounds per square inch absolute, inlet velocity from 0.5 to 11.0 feet per second, and inlet subcooling from 12° to 51° R.

Comparison of upward- and downward-flow data showed that under certain conditions the critical heat flux for downward flow was significantly lower than that for upward flow. Explanations for this difference are made in terms of the relative velocity between the liquid and the vapor phases as influenced by buoyancy. All the data were separable into buoyancy-dependent and buoyancy-independent zones. An annular-dispersed type of flow can be subject to buoyancy while a slug or bubbly type of flow is not, insofar as the critical heat flux is concerned.

The position of the transition from nucleate to film boiling in the test section was dependent on velocity, subcooling, pressure, and buoyancy. The effect of these parameters on the accumulation of vapor precipitated the transition.

For upward flow, a unique reversal in the trend of the critical heat flux with pressure and subcooling was observed above a pressure of 150 pounds per square inch absolute that occurs at fluid inlet velocities below 5 feet per second. At these low velocities, the flowing system behaves like a pool (nonflow) system.

INTRODUCTION

Critical heat fluxes for pool (nonflow) systems are subject to buoyancy forces that result from reduced- and multi-g fields relative to Earth gravity. Reference 3 includes a good bibliography of reduced-g studies, while references 4 and 5 are examples of the many multi-g studies. Since experimental evidence exists that shows the sensitivity of pool systems to gravity forces, it is reasonable to expect that the critical heat fluxes of flow systems should also be subject to buoyancy forces, at least to some limiting fluid velocity.

In most cases, the buoyancy force itself has not been considered a significant parameter. An exception is given in reference 2, which presents an equation that correlates horizontal and vertical water flow in tube bundles for data obtained at 1000 pounds per square inch absolute. The change in orientation from vertical upward to horizontal flow reportedly reduces the critical heat flux significantly.

This investigation was conducted to determine the conditions under which buoyancy influences the critical heat flux of a vertically flowing two-phase heat-transfer system. The critical heat flux is herein defined as the heat flux required to cause a temperature excursion on the wall of a test section resulting in vapor blanketing of the entire surface or a portion of it. A change in the buoyancy force on the fluid was effected by a change in the flow direction. The apparatus consisted of an instrumented test section in a liquid-nitrogen flow system that was capable of being oriented so that fluid flow through the test section would be either in the direction of or opposed to the Earth gravity vector.

Liquid-nitrogen critical heat-flux data were obtained in both flow directions through a resistance-heated, instrumented test section with a 0.505-inch inside diameter and a 12-inch heated length. The test conditions included a system pressure range from 50 to 240 pounds per square inch absolute, an inlet velocity range from 0.5 to 11.0 feet per second, and an inlet subcooling (saturation minus bulk temperature) range from approximately 12° to 51° R.

APPARATUS

Flow System

The cryogenic fluid flow system was composed of three integral units, as shown schematically in figure 1. The three units included a gaseous-helium, high-pressure storage tank mounted in an insulated tank, a 40-gallon liquid-nitrogen supply Dewar, and a vacuum tank containing the test section, mixing chambers, and flow venturi.

The flow system was connected through a system of valves and regulators. The helium gas was used to pressurize the liquid-nitrogen Dewar. A liquid-nitrogen bath for the helium tank precooled the gas to essentially liquid-nitrogen temperatures in order to minimize heat transfer between the gas and the bulk liquid. Pressurization of the Dewar forced the liquid nitrogen to flow through an instrumented, resistance-heated test section

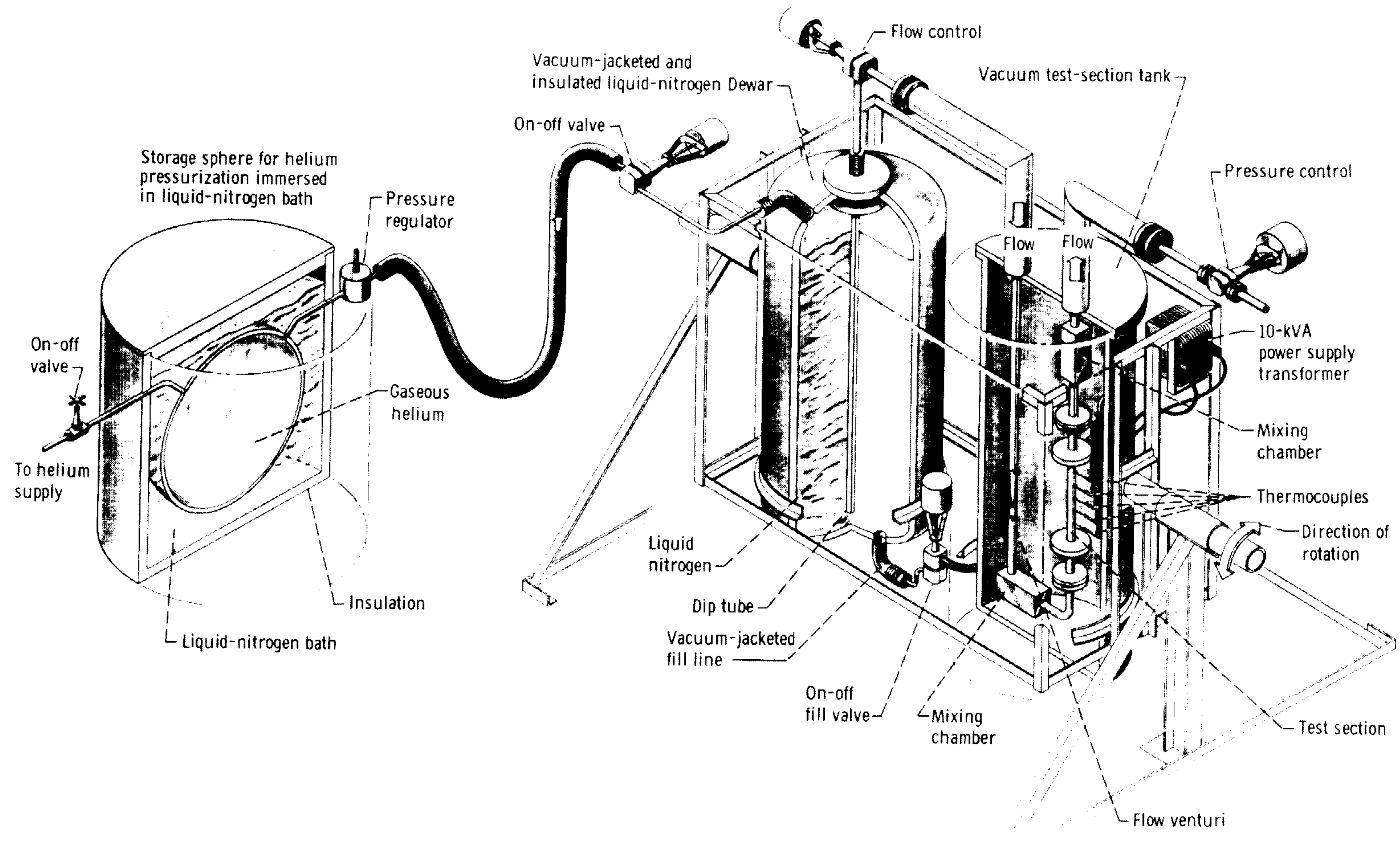


Figure 1. - Liquid-nitrogen heat-transfer apparatus.

that was part of the flow system. Mixing chambers consisting of a system of baffles were located before and after the test section to eliminate temperature stratification in the bulk fluid. The nitrogen was then discharged into the atmosphere.

Flow control was achieved by a throttling valve located at the inlet side of the test-section tank, and pressure control was attained by a throttling valve on the outlet side. The entire flow system up to the pressure control valve was vacuum-jacketed to minimize heat transfer to the cryogenic fluid.

Both the liquid-nitrogen Dewar and the test-section tank were installed in an angle-iron cage that was mounted on trunnion-type supports to allow rotation of the cage. The helium supply tank in the liquid-nitrogen bath was coupled to the Dewar by means of a flexible high-pressure hose. This arrangement made it possible to turn the test section over without changing the flow system. The heat-transfer assembly with the trunnion-mounted cage containing the Dewar and the test-section is shown in figure 2.

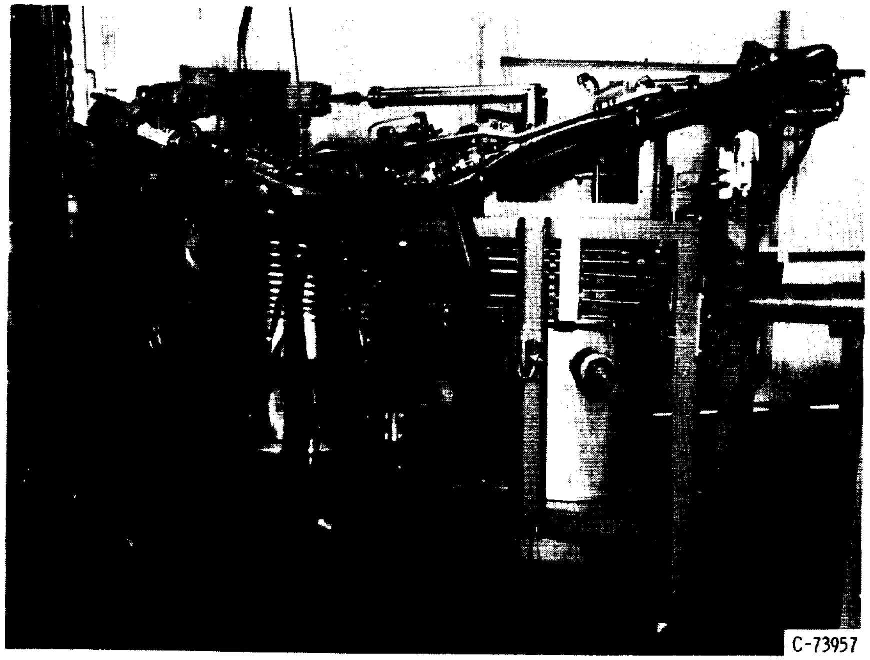
Test Section

The test section was made from nickel-alloy tubing with a 0.505-inch inside diameter and a 0.010-inch wall thickness. Figure 3(a) is a schematic drawing of the test section showing instrumentation stations for wall-temperature and voltage-drop measurements. The resistance-heated portion of the test section was 12 inches long. The thermocouples, voltage taps, and two copper electrodes were silver-soldered to the test section at the indicated locations.

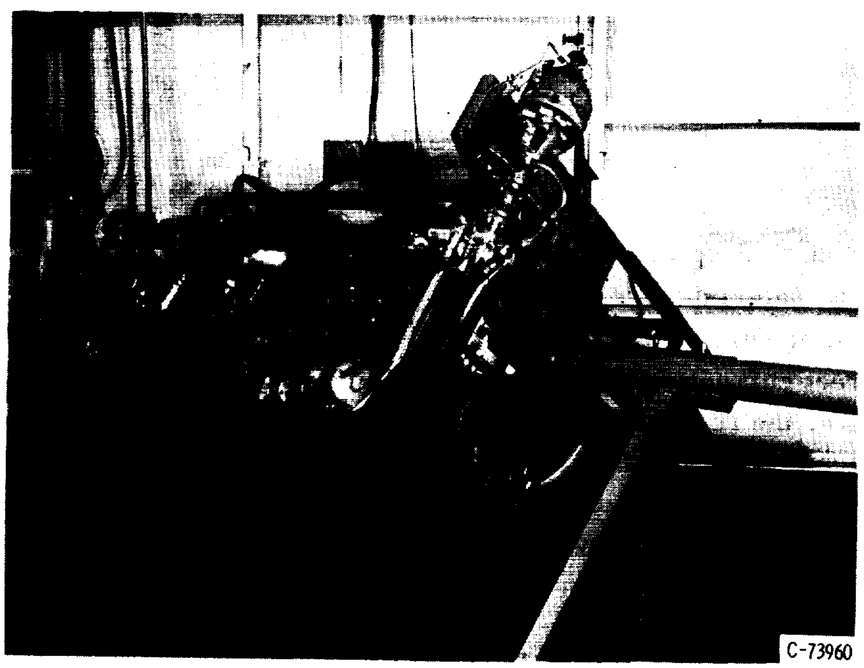
A 10 000-watt, 400-cycle alternator supplied the power to heat the test section. The power input was controlled by a variable transformer to deliver a maximum of 16 volts and 600 amperes. The test section was electrically isolated from the rest of the flow system through insulated connecting flanges.

Instrumentation

The vertical orientation of the test section mounted within the vacuum tank is shown schematically in figure 3(b). Twelve copper-Constantan thermocouples made of 28-gage wire were silver-soldered in two rows 180° apart along the length of the tube. Five voltage taps made of 28-gage copper wire were equally spaced along the tube. Measurements obtained from these taps verified the linearity of the voltage drop. Fluid bulk temperature measurements were made in the mixing chambers located at the inlet and the outlet ends of the test section. Platinum resistance thermometers were used for these measurements. System pressure was measured in the two mixing chambers, and no



(a) Apparatus oriented for upward flow.



(b) Apparatus rotated on trunnion-type mounting.

Figure 2. - Position variation for liquid-nitrogen heat-transfer apparatus.

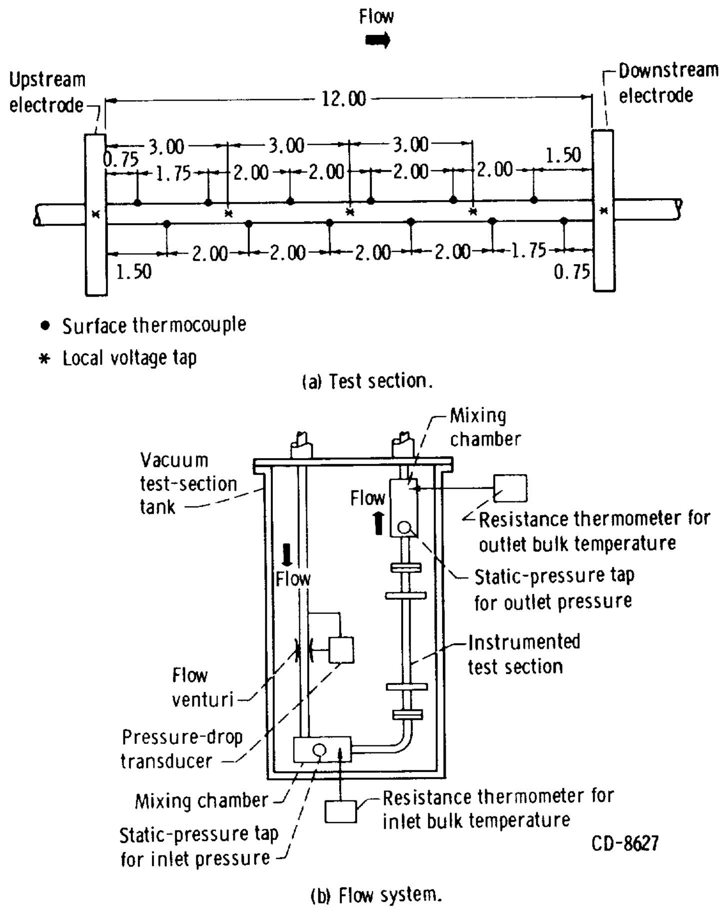


Figure 3. - Instrumentation. (All dimensions are in inches.)

appreciable pressure drop was observed across the test section. Fluid flow rate was measured with a venturi meter.

The alternating current and voltages for heating the test section were converted to low-voltage direct current. These millivolt signals, along with the outputs of the pressure transducers, the resistance thermometers, and the thermocouples, were fed to an automatic voltage digitizer (ref. 6) and recorded on tape.

Before going to the voltage digitizer, the millivolt signals were split and sent to a multichannel oscillograph for visual monitoring of the test during the data runs.

PROCEDURE

The cage containing the Dewar and the test-section tank was mounted on the trunnion supports with the test section oriented to permit upward fluid flow. The buoyancy force in this orientation was in the direction of fluid flow. The experimental procedure then required periods of time for instrument calibration, apparatus cooldown, and, finally, data acquisition.

The cooldown period was initiated by filling the helium tank bath and the 40-gallon Dewar with liquid nitrogen. The Dewar was vented through the test section to precool the entire flow system. Auxiliary thermocouples (not shown in fig. 2(a)) mounted on the electrodes of the test section were used as a cooldown guide. These thermocouples were read out on a strip chart and continuously monitored during the runs. This procedure was necessary because the electrodes, having a large mass compared with the test-section tube, tended to heat up during the power-on and the down-time periods required for refilling the test Dewar. Preliminary data not presented herein showed that hot electrodes could significantly affect the heat-transfer data.

The controlled variables for operating the test apparatus included system pressure, fluid flow rate, and electrical power to heat the test section. A systematic procedure for a data run consisting of fixing a system pressure and obtaining critical heat-flux data over a range of fluid flow rates was then established. At each flow rate the electrical power was turned on and increased by small increments until the condition of criticality was established, at which time the data were recorded and the power cut off. A temperature excursion on the wall of the test section, as viewed on the multichannel oscillograph, was considered the end point of the test. This temperature excursion indicated a transition in the heat-transfer mechanism from nucleate to film boiling. The system pressure was then changed and the running procedure repeated over a range of pressures.

The cage containing the Dewar and test-section tank was then rotated on the trunnion supports so that fluid flow was downward. The buoyancy force in this orientation was in the opposite direction to that of the fluid flow. Removal of the dip stick from the Dewar was required to permit the fluid to flow through the system. The test conditions for the upward-flow data previously obtained were repeated and the data recorded. Since the only difference in test conditions downstream of the inlet control valve for the two sets of data was the direction of fluid flow, any differences in the heat-transfer data had to reflect this change in buoyancy force on the fluid.

The range of critical heat-flux data included system pressures from 50 to 240 pounds per square inch absolute, inlet velocities between 0.5 and 11 feet per second, and inlet subcooling between 12° and 51° R.

For several runs, the subcooling was changed by bubbling warm nitrogen gas through the liquid nitrogen with the Dewar at elevated pressure in order to separate the subcooling effect from the pressure effect on the critical heat-flux data. For all the runs, a pressure drop of about 50 pounds per square inch was maintained across the flow control valve on the inlet side of the test section. The multichannel oscillograph was used to assure stable flow and pressure conditions visually during the tests. In both upward and downward flow, these conditions appeared to be similar. The estimated possible maximum errors in the data are as follows: heat flux, 10 percent; flow rate, 5 percent; system pressure, 3 percent; and inlet bulk temperature, 2 percent.

DATA PRESENTATION

Upward- and downward-flow heat-transfer data comparisons are presented in figure 4. Figure 4(a) shows typical data obtained at a system pressure of 50 pounds per square inch absolute and an inlet subcooling of 12° R. The average deviation of the mean subcooling value as shown on this and succeeding plots is between 3.5° and 4.0° R. Upward-flow data appear to be independent of inlet velocity over the range covered. Downward-flow data are velocity dependent in the zones from 0.5 to 2.5 feet per second and from 8 to 10.5 feet per second.

The difference in the level of the critical heat flux as defined in the INTRODUCTION is caused by the change in direction of the buoyancy force on the flow system. Between inlet velocities of 2.5 and 8 feet per second, a change of direction from upward flow to downward flow results in a decrease of about 22 percent in the critical heat flux. Below 2.5 feet per second, there is an increased dropoff to 86 percent of upward-flow value at an inlet velocity of about 1 foot per second. Above 8 feet per second, downward-flow data approach upward-flow data and finally both merge at 10.5 feet per second. The transition from nucleate to film boiling occurs at the outlet end of the test section for upward flow. For downward flow below 2.5 feet per second, the data represented by the triangles describe transition at the inlet or at any position other than the outlet end of the test section.

Figure 4(b) compares data for a system pressure of 75 pounds per square inch absolute and an inlet subcooling of 19° R. In this case, the upward-flow data are independent of inlet velocity up to about 7 feet per second, but above this velocity the slope of the data curve increases. Above 7 feet per second, the critical heat flux appears to be independent of flow direction because the upward- and downward-flow data merge into a common curve. The slope and the position of downward-flow data coincide with those for upward-flow data but extend down to a critical heat-flux value of 0.10 Btu per second per square inch at 2 feet per second, at which position a more rapid dropoff in heat flux is observed. The occurrence of the temperature excursion at positions other than the outlet end of the test section again evidences itself below about 2 feet per second.

Figures 4(c) to (f) are the basic data plots for system pressures of 100, 150, 200, and 240 pounds per square inch absolute at inlet subcooling values of 26°, 35°, 43°, and 51° R, respectively. A decreasing buoyancy effect is apparent on the critical heat flux with increased pressures and subcooling. In all cases, the point of intersection of the upward- and downward-flow curves represents the minimum inlet velocity for that set of

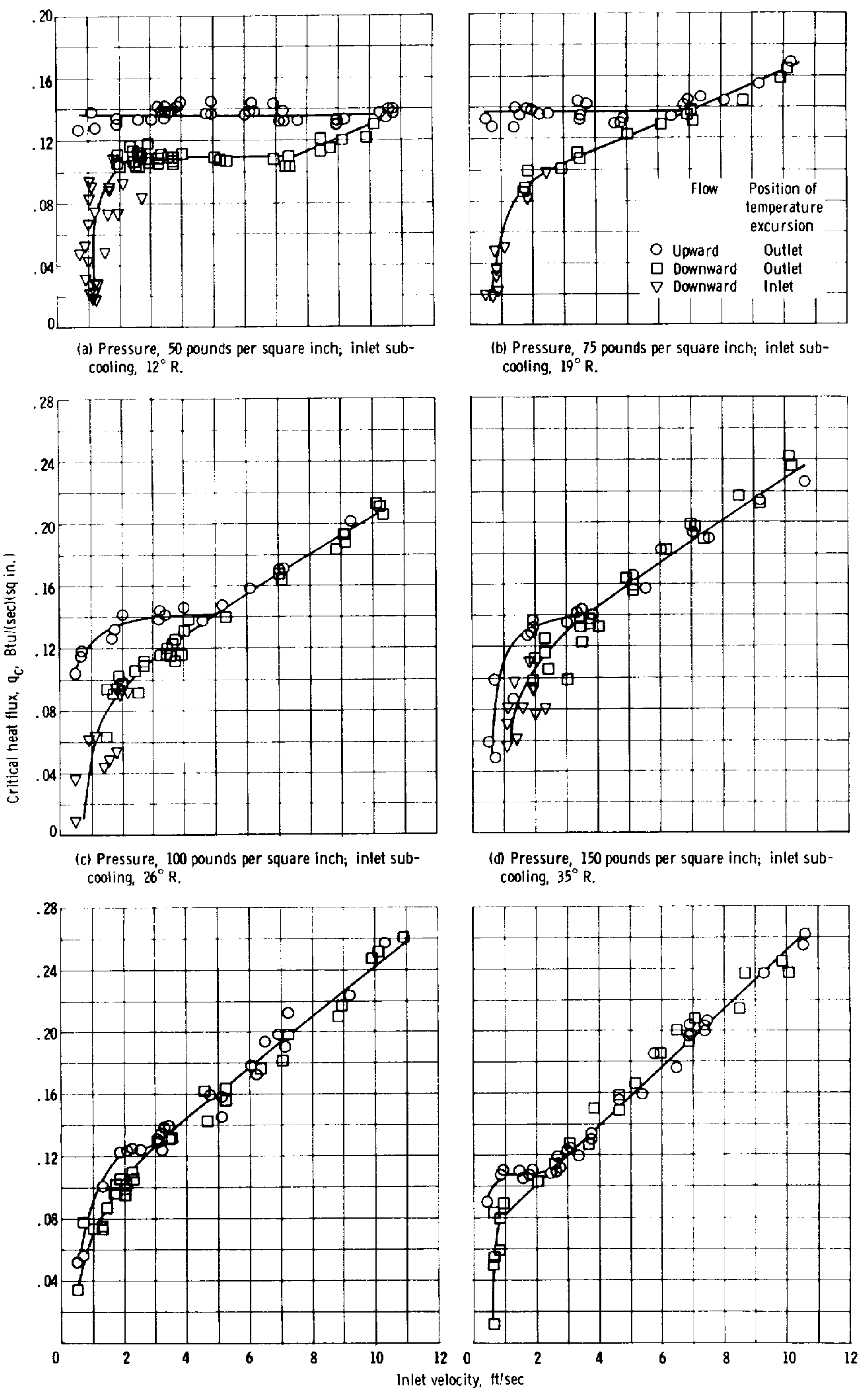
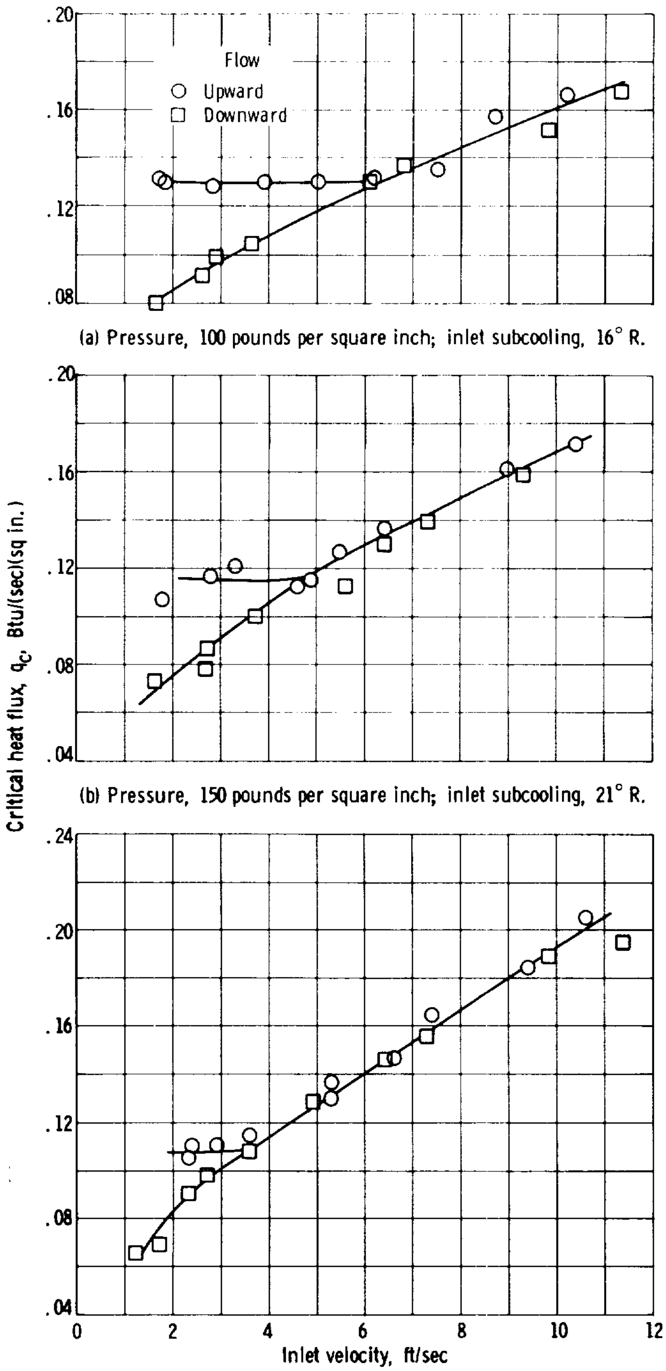


Figure 4. - Critical heat flux as function of inlet velocity for upward and downward flow of liquid nitrogen.



(c) Pressure, 200 pounds per square inch; inlet subcooling, 25° R.

Figure 5. - Critical heat flux as function of inlet velocity for upward and downward flow of liquid nitrogen.

conditions above which the fluid momentum is sufficient to negate any buoyancy effects.

All the basic data were obtained with a constant inlet bulk temperature of about 145° R. A limited amount of data was also obtained at values of subcooling other than those presented in the basic data. The effect of varying the subcooling at constant pressure was examined for three pressure levels. Figure 5 presents the data for system pressures of 100, 150, and 200 pounds per square inch absolute with inlet subcooling temperatures of 16°, 21°, and 25° R, respectively.

DISCUSSION OF RESULTS

Buoyancy Against Critical Heat Flux

As shown in figure 4, the inlet velocity at which the upward- and downward-flow data merge is sensitive to the system pressure and inlet subcooling. At 50 pounds per square inch absolute and 12° R subcooling (fig. 4(a)), the point of intersection is at an inlet velocity of 10 feet per second. Below an inlet velocity of 10 feet per second, a system operating under this set of conditions and geometry would be influenced by changes in buoyancy. As the pressure and subcooling are increased (fig. 4(b) to (f)), the position of intersection of the curves shifts to lower values of inlet velocity. Increasing the pressure and subcooling therefore decreases the velocity below which buoyancy effects on the critical heat flux, as defined herein, can be observed.

The points of intersection of all data curves (figs. 4 and 5) were replotted in figure 6 with coordinates of inlet subcooling ($^{\circ}$R) and inlet velocity (ft/sec). The solid line connects data from 50 to 240 pounds per square inch absolute obtained with the fluid entering the test section at an inlet bulk temperature of about 145° R. The dashed lines represent data obtained at a fixed system pressure but at different inlet subcooling. Inlet velocities less than those represented by the data points fall into a buoyancy-dependent zone, while higher velocities fall into a buoyancy-independent zone.

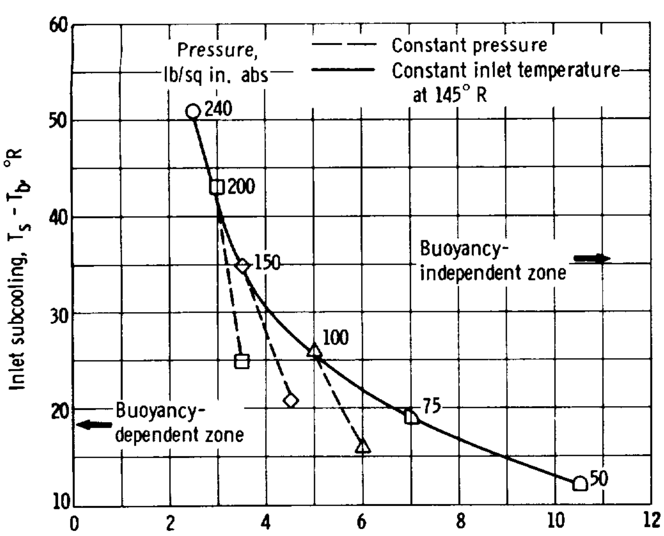


Figure 6. - Susceptibility of liquid nitrogen flow system to buoyant effects on critical heat flux.

The sharp separation in the data implies an abrupt change in the flow pattern from one that is susceptible to buoyancy forces to one that is not. It is suggested that an annular-dispersed flow model characterized by liquid droplets in a gaseous core surrounded by a liquid annulus on the wall of the test section exists in the buoyancy-dependent zone. In this type of flow model, the thickness of the liquid layer on the wall can be affected by changes in buoyancy. For the same flow rate, downward flow should result in a thinner liquid layer than upward flow because the relative velocity (slip) between the liquid and the vapor is lower for the downward flow. The lower vapor velocity results in an increased vapor accumulation. A thinner liquid layer presents less protection for the wall and thereby results in lower values of critical heat flux than those observed for upward flow. An abrupt change in the model from annular-dispersed to slug or bubbly flow is then speculated for the data that fall in the buoyancy-independent zone. The critical heat fluxes in these cases are controlled by a mechanism that is not significantly influenced by changes in buoyancy.

An extrapolation of the data in figure 6 would indicate that a family of constant-pressure curves could exist as an extension of the dashed lines. The upper limit of the curves would occur for subcooling when boiling ceases or the bubbles collapse on the heat-transfer surface without departing. Such a condition could exist when the temperature of the liquid nitrogen is near the freezing point. The lower limit of the family of curves should occur on the zero subcooling line at some finite inlet velocity. The point of intersection would be dependent on the pressure level, with decreasing pressures resulting in higher velocities. The apparatus and test geometry used for this investigation limited maximum inlet velocities to about 11 feet per second, which was not of sufficient magnitude to determine the end points of the curves. Data curves (not presented herein) with fluid entering the test section at saturated conditions would not converge at the maximum inlet velocities obtained.

Figure 6 is presented to separate the data into two distinct zones. The buoyancy-independent zone is explicit in that data obtained under these test conditions would not be influenced by buoyancy effects. The test conditions in the buoyancy-dependent zone, on the other hand, are subject to buoyancy effects, but the magnitude of these effects cannot be obtained from the plot. The data comparisons shown in figures 4 and 5 (pp. 9 and 10, respectively) should be used for this purpose, and it should be emphasized that the data are limited to a particular fluid and test-section geometry.

Velocity Against Critical Heat Flux

Upward flow. - Upward-flow data comparisons relating critical heat flux, inlet velocity, pressure, and inlet subcooling temperature were facilitated by replotting the basic data curves of figure 4 (p. 9) into a single plot (fig. 7(a)). Several trends in the data can be readily observed.

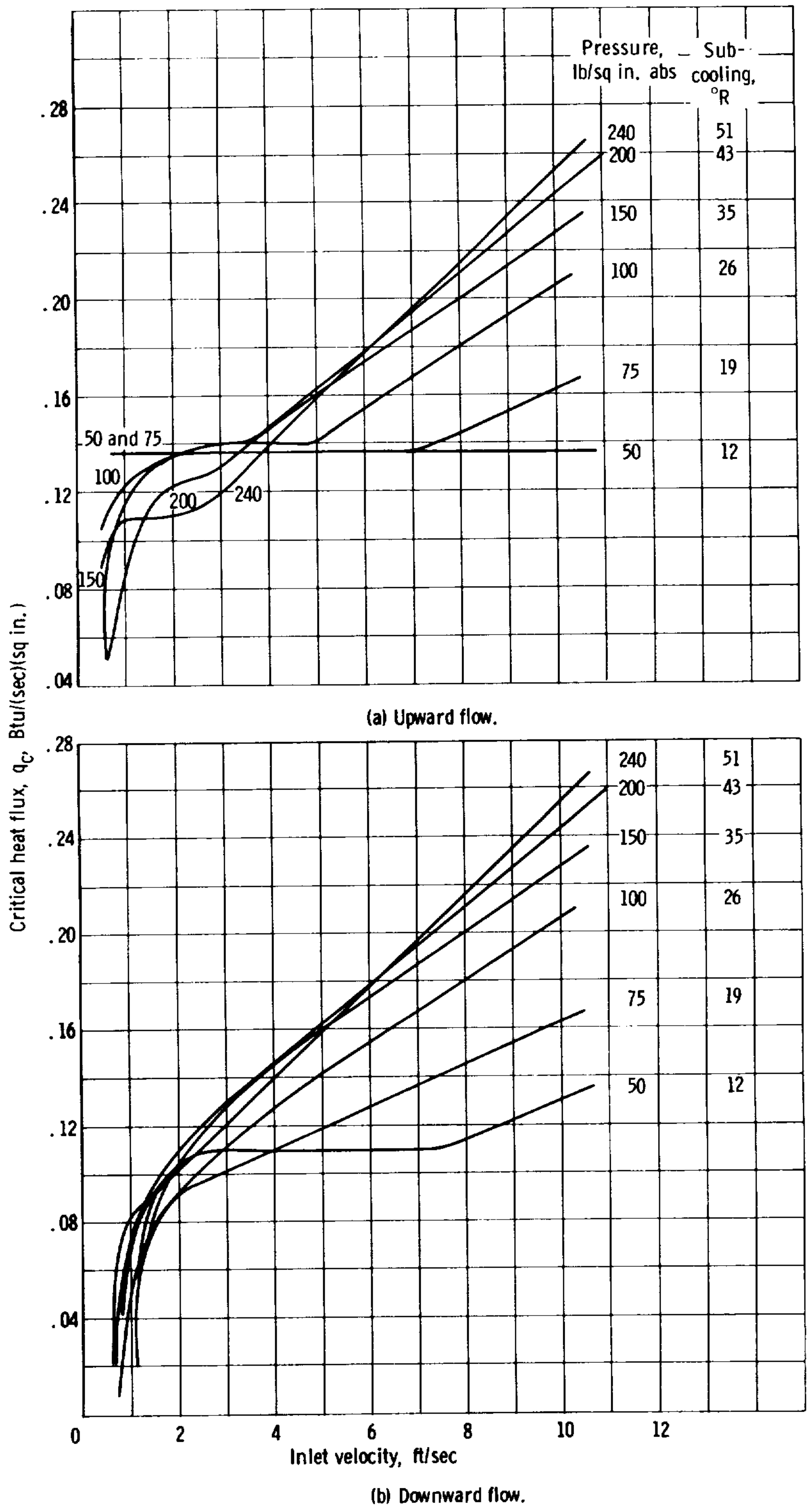


Figure 7. - Critical heat flux and inlet velocity as function of pressure and subcooling for liquid nitrogen in round tubes.

The first trend is the apparent absence of inlet velocity effects on the critical heat flux for the lower pressure and subcooling data along all or a portion of the data curves. The curve representing the 50-pound-per-square-inch-absolute data is essentially a straight horizontal line showing a critical heat-flux value independent of changes in inlet velocity, at least up to the maximum inlet velocity obtained. The 75-pound-per-square-inch-absolute curve is horizontal up to an inlet velocity of 7 feet per second, above which the critical heat flux increases with velocity. At 100 pounds per square inch absolute, the position of velocity dependence moves to about 5 feet per second.

The insensitivity of changes in the critical heat flux with inlet velocity for portions of some of the data curves could be explained by some simple calculations made for the 50-pound-per-square-inch-absolute data. The velocity at the exit, where the transition from nucleate to film boiling takes place, is nearly a constant, even though the inlet velocity varied from 0.5 to 10.5 feet per second. An average density was used with quality based on equilibrium conditions and no slip between the liquid and the vapor. The balance of the data that show velocity effects on the critical heat flux could possibly be explained by increased exit velocities that cannot be calculated without making gross assumptions.

Above 150 pounds per square inch absolute, a rather unique trend in forced-flow boiling data is observed. For fixed inlet velocities above approximately 5 feet per second, the critical heat flux increases with increasing pressure and subcooling. At velocities below 5 feet per second, there is a reversal in the trend of the critical heat flux; an increase of pressure and subcooling lowers the critical heat flux. This trend is consistent with pool (nonflow) boiling data presented in the literature (ref. 7), which show the existence of a maximum critical heat flux at one-third of the critical pressure. At higher pressures, the drop in the critical heat flux can be attributed to a lower heat of vaporization with increased pressure. The low-velocity forced-flow boiling data for pressures above 150 pounds per square inch absolute appear to behave like pool data.

Downward flow. - Marked differences between downward-flow (fig. 7(b)) and upward-flow data (fig. 7(a)) are apparent. Regions that do not show a velocity dependence on the critical heat flux exist only for a portion of the 50-pound-per-square-inch-absolute curve. The rest of the data, including the low pressure and subcooling data, shows an increase in critical heat flux with increased inlet velocity. Above 2 feet per second, the critical heat flux generally increases with pressure and subcooling. Below 2 feet per second, the data merge and there is a rapid increase in slope. Vapor accumulation causes this temperature excursion to occur at a position other than the outlet end of the test section.

The trend reversal in the critical heat flux with high pressures and subcooling at low inlet velocities as seen in upward-flow data (fig. 7(a)) does not appear in downward-flow data. The change in direction of the buoyancy force in some way offsets the drop in the heat of vaporization that accompanies higher pressures.

Position of Transition From Nucleate to Film Boiling

In defining the critical heat flux as the heat flux required to cause a temperature excursion on the wall of the test section, it was necessary to specify that this transition from nucleate to film boiling would occur at any position on the tube. For upward flow, the temperature excursion always occurred at the outlet end of the tube; this behavior is consistent with that which is generally believed (ref. 1) to be true. On the other hand, for downward flow, this transition could occur at the inlet, the outlet, or at any other position in the tube. An examination of the basic data (fig. 4, p. 9) shows that the position of the temperature excursion is pressure, subcooling, and velocity dependent.

Downward-flow data are divided into two groups: critical-heat-flux data with the temperature excursion at the outlet end of the test section, and data with transition at the inlet or at any position other than the outlet. The velocity dependence of the transition position is apparent from figures 4(a) to (d). For inlet velocities below about 2 feet per second, the temperature excursion generally starts at a position other than the outlet, while above 2 feet per second the transition is at the outlet end of the tube.

For pressures above 150 pounds per square inch absolute and inlet subcooling above 35° R (figs. 4(e) and (f)), the transition always occurs at the outlet end regardless of inlet velocities. Apparently, the higher pressures and subcooling so limit the size of the bubbles that they are not strongly influenced by changes in the buoyancy force.

In both upward and downward flow, the accumulation of vapor is considered the local condition that causes the temperature excursion to occur at the various positions in the tube. For the upward flow, the buoyancy force enables the vapor to move faster than the liquid. Consequently, the vapor accumulates at the outlet end of the tube, the result being a maximum void distribution that precipitates the temperature excursion. In downward flow, the buoyancy force so acts in opposition to the liquid flow that the relative velocity between the vapor and the liquid is less than in upward flow. The more rapid accumulation of vapor results in a premature transition from nucleate to film boiling. At inlet velocities below approximately 2 feet per second, the vapor could rise against the direction of flow and accumulate at the inlet end of the tube. The motion picture supplement to reference 8, which is a visual study of low-velocity boiling of liquid nitrogen in upward and downward flow, substantiates this observation.

The level of the critical heat flux at these low inlet velocities depends on the position of transition within the tube. The lowest critical heat flux occurs with transition at the inlet and increases as transition moves downstream to the outlet. Increased scatter in the basic data below 2 feet per second could be attributed to the statistical nature of bubble agglomeration and the effect of flow turbulence on the resulting vapor masses.

CONCLUSIONS

An investigation was conducted of the buoyancy effects on the critical heat flux of forced convective boiling in vertical flow. Liquid nitrogen was the working medium flowing through a 0.505-inch-inside-diameter by 12-inch-long tube subject to uniform heat fluxes. Critical heat flux data were obtained with the flow direction upward and downward over a range of test conditions that included system pressures from 50 to 240 pounds per square inch absolute, inlet velocities from 0.5 to 11.0 feet per second, and inlet subcooling from 12° to 51° R.

1. Comparison of upward- and downward-flow data showed that under certain conditions the critical heat flux for the downward flow was significantly lower than that for the upward flow.

2. Results suggest that the critical heat flux is subject to buoyancy forces when the test conditions produce an annular-dispersed type of flow. This is not so when slug or bubbly flow persists.

3. The buoyancy effects on the critical heat flux increased as the pressure and subcooling decreased and decreased as the velocity increased. Above certain velocities, depending on pressure and subcooling, the buoyancy effect was eliminated by the momentum of the fluid.

4. The position of the transition from nucleate to film boiling was velocity, subcooling, pressure, and buoyancy dependent. In upward flow, the temperature excursion always started at the outlet end of the test section regardless of test conditions. In downward flow, the temperature excursion started at the inlet, the outlet, or at any other position in the tube, depending on an interaction of the system variables. Above 150 pounds per square inch absolute and subcooling of 35° R, the transition always started at the outlet. At lower pressures and subcooling, the position was velocity dependent. Above approximately 2 feet per second, the transition occurred at the outlet, but below that inlet velocity it occurred at the inlet or at any position other than the outlet. The critical heat flux decreased as the transition point moved closer to the inlet side of the tube.

5. Vapor accumulation was considered the local condition that caused the temperature excursion to occur at the different positions in the tube. In upward flow, the vapor moved faster than the liquid because of the direction of the buoyancy force; the maximum vapor accumulation was therefore at the outlet. The vapor velocity was lower in downward flow, and therefore vapor accumulated faster. At velocities below 2 feet per second, the predominance of the buoyancy force could cause the bubbles to rise against the flow and accumulate at the inlet.

6. For upward flow, a unique reversal in the trend of the critical heat flux was observed with pressure above 150 pounds per square inch absolute and subcooling of 35° R. Above an inlet velocity of about 5 feet per second, an increase of pressure and subcooling

increases the critical heat flux, while below 5 feet per second there is a decrease in the critical heat flux. At the low velocities, the flowing system behaves like a pool (nonflow) system. A decrease in the critical heat flux above specific pressures in pool systems is substantiated by the reference literature.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, August 3, 1966,
129-01-09-04-22.

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